IBM II (B)

Reliability Improvement of a Geneva Mechanism

Chuck explained how he proceeded with his analysis. "Before attempting any mathematical development, I wanted to test the geneva mechanism to get some idea of the dynamic forces involved. I felt it would be helpful to know what kind of answers to expect. We felt mathematical analysis should be used to speed the project along, but experimental measurements would give us something to compare our theoretical predictions with. If there had been no limit on the amount of time available we would have done much more testing. However, I don't think we'd have ever attempted a purely 'cut and try' solution because this assumes that one knows very little about the problem to begin with. Such an approach seems to indicate that the engineer sees no correlation between the various parameters of the problem, such as contact stress and wear." In an IBM publication entitled Handbook of Metal Wear Properties, Chuck believed he had found justification for his feeling that, all other factors being equal, wear is proportional to contact stress.

Direct Measurement of Force between Drive Pin and Star Wheel Slot

"Since I was concerned with contact stresses between drive pin and star wheel slot, it seemed logical to measure the load on the pin during an operating cycle. About the only way to measure pin loads would be to locate a strain gauge on the drive wheel between the pin and the shaft and record the strains. These strains would be directly proportional to the pin forces. But since the drive wheel rotates we'd have to use slip rings in the voltage sensing circuit, and we were afraid the slip rings would introduce a lot of noise into the signal from the strain gauge.

"There didn't seem to be an alternative to using slip rings. Consequently, we looked for ways to increase the signal so the noise would be less of a problem. The obvious way was to increase deflection of the pin under load, so we relieved the driving wheel by machining to reduce thickness of the material between the pin and the center of the wheel. (See Exhibit 1. This mechanism is a later design than that shown previously in Exhibit 4 of Part A.) The portions of the wheel supporting the pins were then attached to the center of the drive wheel only by narrow, weak sections resembling cantilevered beams. We attached a strain gauge on the relieved surface at what might be called the 'top' of one of the beams and calibrated the experimental setup by hanging known weights on the pin. Then, when the mechanism was driven at its design speed of 4000 rpm, the strain gauge output gave a record of the pin load which we put on magnetic tape. We recorded the tape at 120 inches per second, the recorder's fastest speed: then played it back as slowly as possible (15 IPS) and fed

the signal into a strip chart recorder. We had to slow the play-back because the cycle time of the geneva is about 7.5 milliseconds and no chart recorder made could keep up with it."

These tests were performed during January 1964. A typical experimental force-time record is shown in Exhibit 2. The signal from the strain gauge was filtered, after being first recorded, to eliminate two resonant vibrations. The two resonant frequencies are apparent in the unfiltered trace. One occurs when the instrumented pin is in contact with the slot, the other between engagements. If these high frequency vibrations were present only in the relieved driver and not in the unmodified part, Chuck wanted to filter them out so the experimental plot would represent more faithfully the forces on an uninstrumented drive pin. Chuck modeled the case of no contact between pin and slot -- the free pin vibrating at the end of a torsional spring. The mass corresponds to the pin and that part of the driver to which it is attached; the spring corresponds to the thin relieved "beam." Chuck measured the spring constant of the beam and calculated its mass from the size and density of the drive pin and its support. The natural frequency he calculated for this system corresponded to one of those observed on the experimental recordings and this frequency was then filtered out. A similar calculation for a mass on a spring in contact with another mass, representing the star wheel, gave a frequency corresponding to the second observed resonant frequency which he also filtered out.

"The load on the pin varies in both magnitude and direction throughout a cycle -- from the time the pin enters the slot until it leaves," Chuck stated. "When the pin enters the slot, the load is nearly perpendicular to this axis. That is, our 'beam' is now essentially in bending. When the pin reaches the end of the slot, at the point where it reverses direction, the load is parallel to the beam axis and mainly axial. Since the direction of the pin load varies during the cycle, most of the time we were only measuring part of it."

Photoelastic Testing

Chuck saw a number of alternative ways to reduce contact stress between drive pin and slot surface. He could either decrease the force (that is, the pin load) or increase the effective area of contact. Contact area could be increased, for instance, by using larger diameter drive pins or a thicker star wheel. But Chuck felt that the most promising alternative was to reduce pin loads by reducing the system inertia. Removing material from the star wheel seemed a logical way to reduce inertia, but first Chuck thought he should study the distribution of flexural stresses in the wheel so that he could prescribe removal of material only in regions of low stress.

"The next problem," Chuck continued, "was to determine the distribution of flexural stresses in the wheel. It seemed to me that photoelastic tests on an enlarged plastic model of the wheel would be much easier than a theoretical computation of stresses. I consulted the Handbook of

Experimental Stress Analysis, edited by Professor Hetenyi of Stanford University, to find a suitable birefringent plastic for modeling the geneva wheel. Then I phoned our plastics lab and they recommended a material called Homolite 100.

"We chose to make our plastic model about 6-1/2 times larger in diameter than the actual geneva wheel to facilitate machining of the model and to simplify observation of the induced stress patterns. The back surface of the plastic model was painted with reflective paint so that we could use our lab's small hand-held reflecting polariscope to observe the stress patterns. We loaded the star wheel model with a lever pivoted about its center. A pin in the lever applied a force to the slot simulating the drive pin load. However, we could not include the frictional force between the slot and the pin during dynamic operation." The dimensions of the model star wheel were:

	STEEL STAR WHEEL	MODEL
Star Wheel Radius, in.	.5663	3.64
Thickness, in.	.125	.267
Lock Radius, in.	.292	1.86
Pin Diameter, in.	.156	1.00

The pholotelastic tests as well as some of the pin load experiments were performed in January. A series of photographs of the fringe patterns in the plastic wheel taken through the polariscope as the load position changed appear in Exhibit 3.

Chuck continued, "The concave locking surfaces of the steel geneva extend beyond the face of the wheel to lock it during the dwell portion of the cycle. To simplify the plastic model, we neglected these raised locking surfaces. We intuitively felt that this would be close to the real situation and we expected that our model would give results on the safe side."

Since the experimentally determined pin loads included only the forces perpendicular to the instrumented "beam axis," they could not be used to test the photoelastic model. Chuck calculated the pin loads that he applied to the model from his knowledge of the kinematics and dynamics of the system by setting the torque applied to the star wheel equal to the time rate of change of angular momentum. When he compared components of the calculated pin loads with the experimental records he found what he believed was a satisfactory correlation.

Chuck said, "We were only interested in the maximum flexural stress occurring within the wheel for any point of force application along the slot. At each load position we gradually increased the force while observing the fringe pattern in the wheel. We could quickly determine the point in the wheel at which stresses were greatest for this particular point of load application. Then we increased the load until exactly

three fringes had passed the point of maximum stress. This is a simple way of doing a photoelastic stress analysis. Our model gave a relation between load and flexural stress at the point of maximum stress. With the known similarity relationship between stresses in the model and those in the actual steel wheel and with the loads we had calculated from our knowledge of system dynamics, we felt we could determine the magnitude and location of the maximum flexural stress in the actual steel geneva wheel at various times during its operating cycle. We had to assume only that the actual three-dimensional stress field could be adequately represented in two dimensions ~~and~~ that edge effects could be neglected."

Exhibit 4 shows Chuck's plot of stress at the point of maximum stress, in the actual steel star wheel as a function of load position. The abscissa, marked RAD., gives load position along the slot surface in terms of star wheel radius to that point. The notations on the data points give the location of the point of maximum stress; they refer to the inset diagram of a portion of the geneva wheel. In all cases, the points of maximum flexural stress were found to be at or near the outer surface of the slot or locking radius; thus, positions could be defined by specifying one of the three surfaces A, B, or C, and the ring number.

By the time the photoelastic tests were completed, about two man-months of engineering time had gone into the project; other costs of the geneva study had, according to Chuck, been "relatively small."

"It seemed to me that we were getting useful data from the experiments," Chuck observed. "But this all was costing the company money. I was spending about half of my time on the project and another engineer under my direction was spending a comparable amount. Questions before me as I reflected on our progress were, 'How are we doing?', 'What should we do next?', and 'How can we derive maximum benefit from these investigations?'"

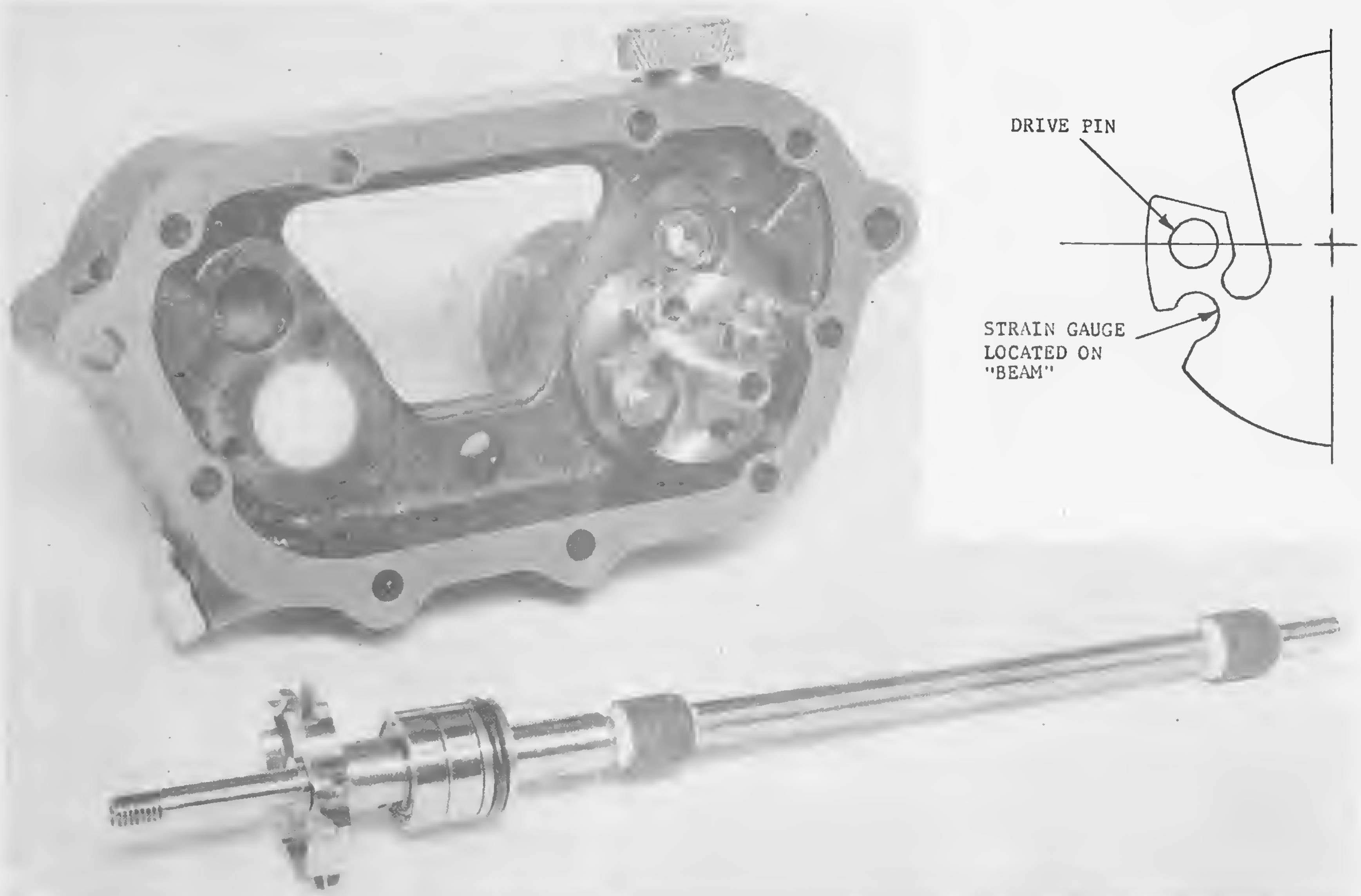


Exhibit 1: Geneva Mechanism with Relieved Drive Wheel for Determination of Pin Loads.

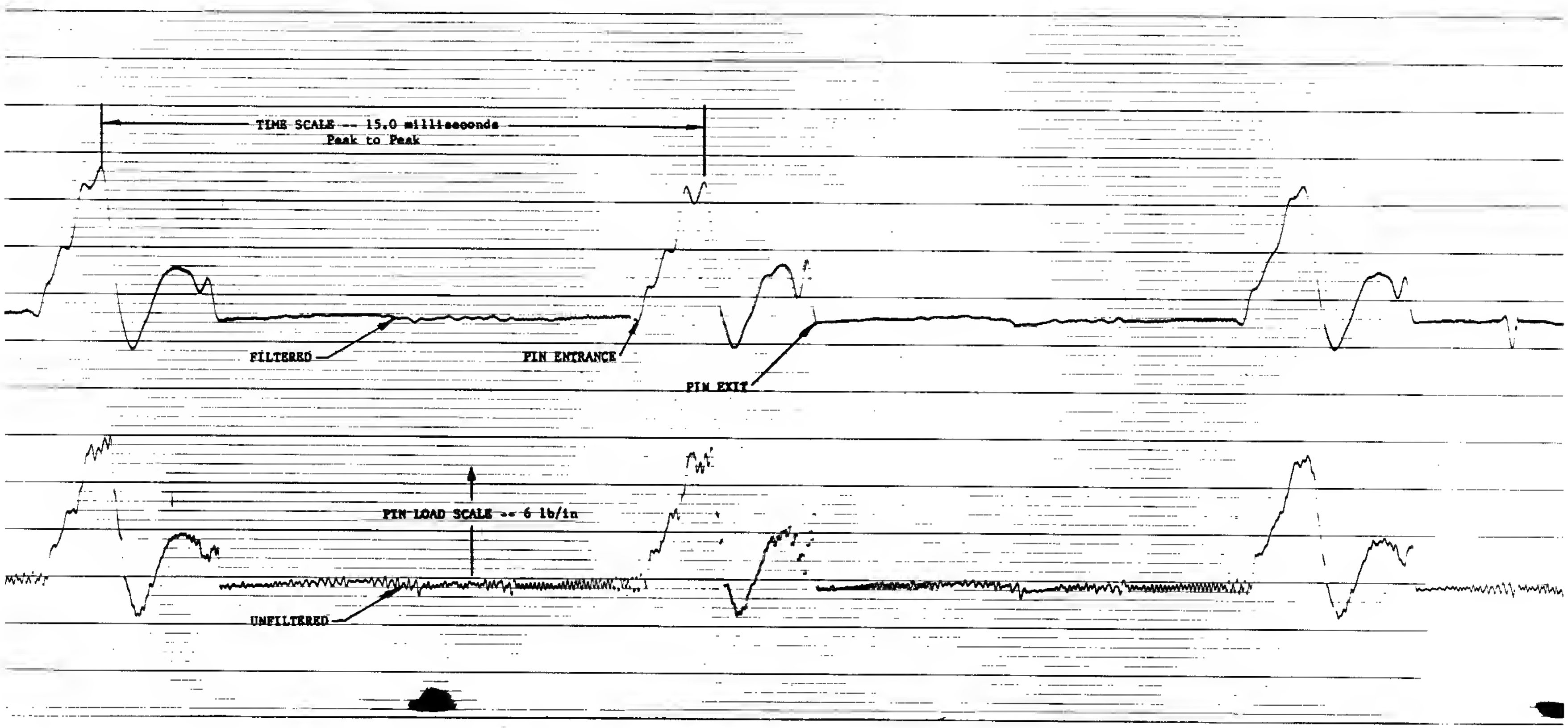
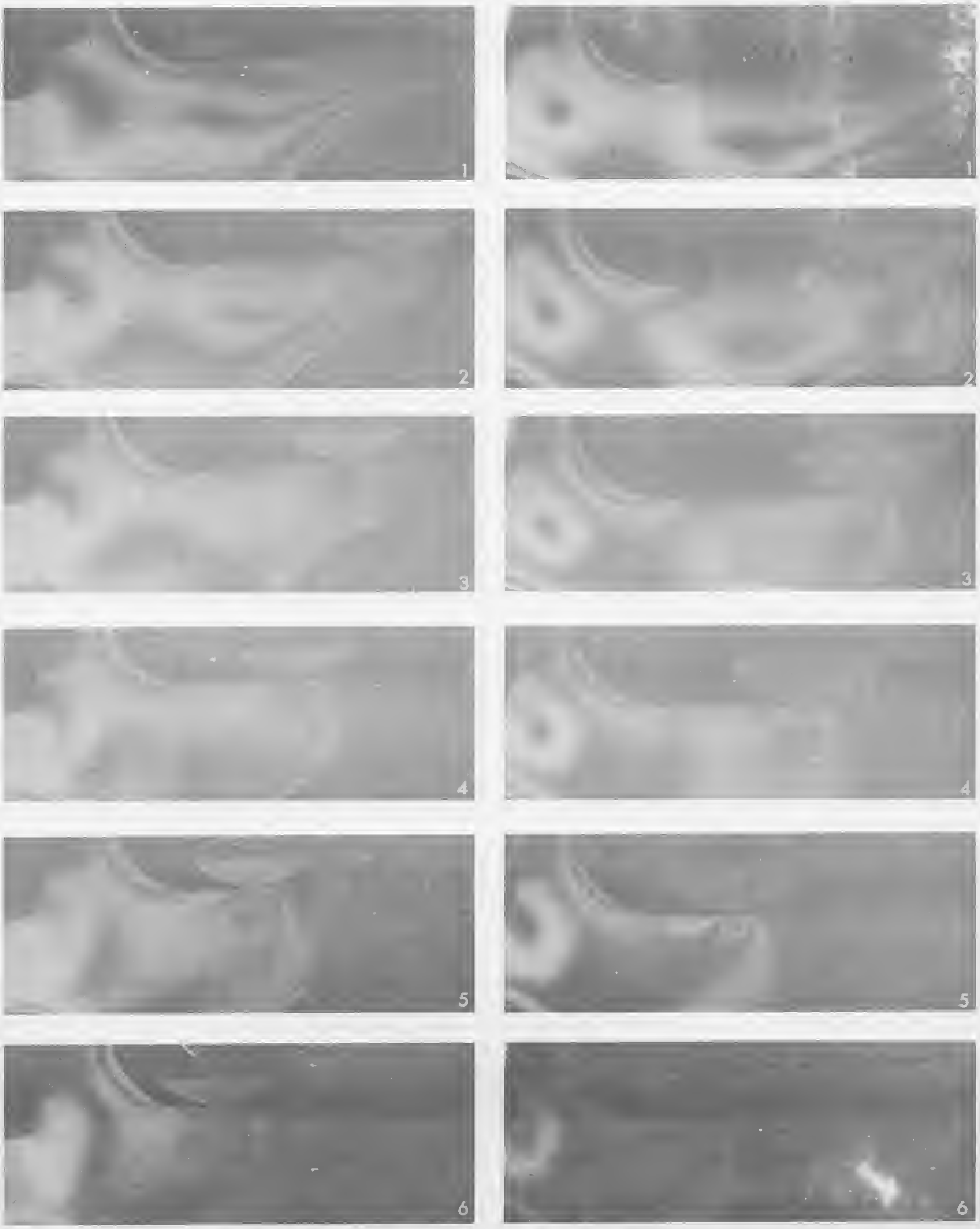
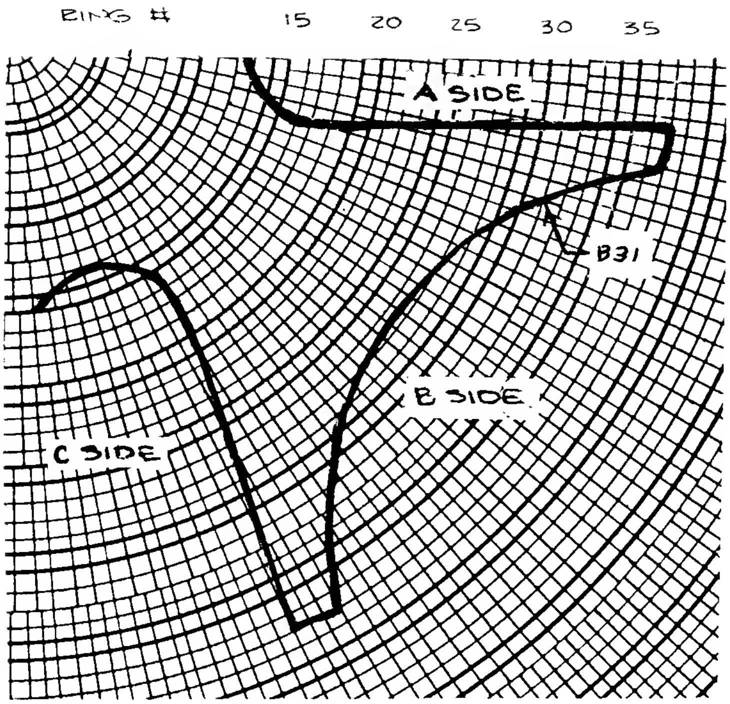


Exhibit 2: Experimental Record of Pin Loads.

<u>Exhibit</u> <u>3:</u> Photoelastic Models of Geneva Wheels of Two Different Proportions with Changing Load Position.





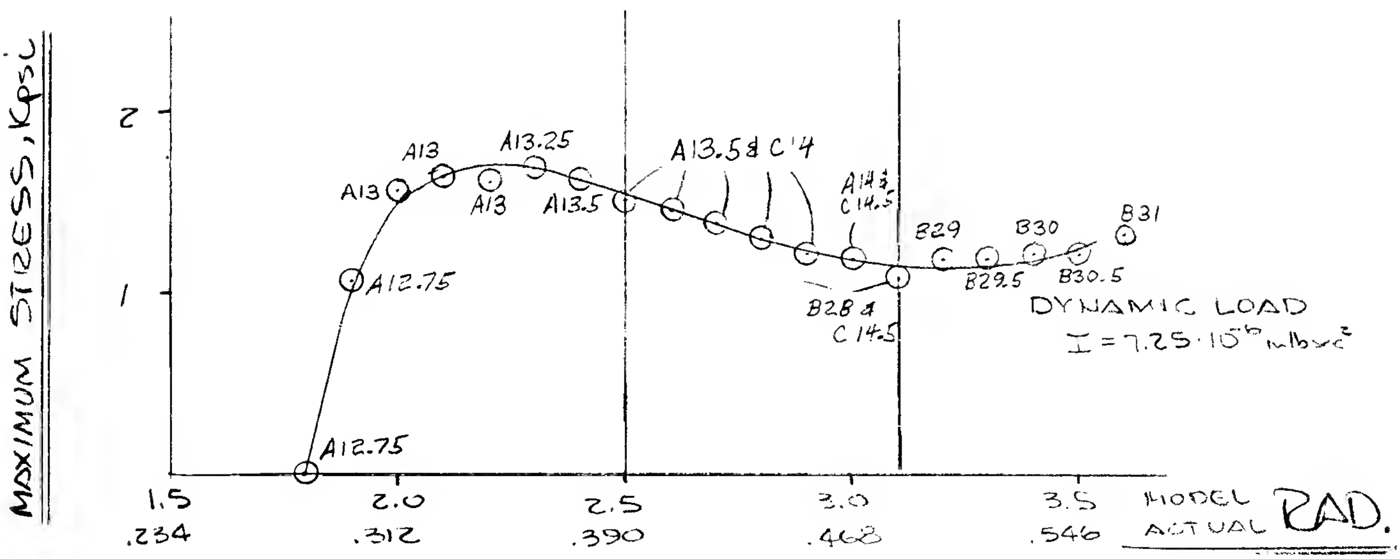


Exhibit 4: Chuck Adams' Plot of Maximum Stress in the Star Wheel as a Function of Load Position. Top Chart Gives Position of Point of Maximum Stress.

IBM II (C)

Reliability Improvement of a Geneva Mechanism

As the photoelastic stress analysis of the geneva wheel progressed, so did Chuck's thinking on how best to continue toward his goal of reducing contact stress between the pin and slot. "I knew," he said, "that while the plastic model would give us a good idea of the stress in the current geneva design, it would be an expensive and time-consuming task to make and test a new plastic model every time we wanted to try to reduce contact stress by changing the shape or dimensions of the star wheel. But we felt it would be necessary to keep an eye on the flexural stress in the wheel. A mathematical method of determining stress is much better from this standpoint; then basic dimensions can be changed at will without going to the trouble of making a model. We did not feel that we could predict the effect of changing parameters by studying the experimental results we already had, but while we were doing the photoelastic study of the model it began to seem that we could calculate stresses in the wheel by using simple beam theory.

"It was mostly just an intuitive feeling, triggered by the appearance of the fringe patterns in our model, that suggested our attempt to treat each segment of the star wheel as a beam. The wheel near the tip of a slot seemed much like a cantilevered beam; the fringe patterns showed a neutral axis. For a force applied near the slot entrance, the neutral axis appeared to bisect approximately the tip region (see Exhibit 1). Near the root of the slot the neutral axis seemed to bisect the angle between two adjacent slots. Assuming these were neutral axes for loads near the tip and the root of the slot, I decided to try calculating stresses using the simple cantilever beam formula, where the stress $\sigma = \frac{My}{I}$.

"The photoelastic tests seemed to show that if the maximum internal stress occurred in the tip region, the corresponding driving pin location would be at the end of the tip. It also seemed that if the maximum internal stress occurred in the root section, at the closed end of the slot, the driving pin would be near the position corresponding to maximum wheel acceleration.

"While we were in the midst of the photoelastic model testing I first saw R. C. Johnson's article in Machine Design on the optimum design of geneva mechanisms (Machine Design, March 22, 1956, page 107). I noticed, however, that Johnson's article dealt only with a single proportional set of genevas -- that is, with a set of geometrically similar wheels. And I thought perhaps genevas with different geometric proportions from that of the 'Johnson wheel' would give different contact stresses. A design based on Johnson's method would only give the optimum for his set of wheels, while there would be sets of genevas with different proportions for which the optimum wheel would have lower contact stresses."

Stress Calculations

Chuck had determined from the photoelastic stress analysis that the maximum flexural stress in the wheel could occur either near the tip of the wheel or near the root of a slot. These correspond to one of two load conditions: angular velocity of the wheel at zero (pin at tip of slot), or angular acceleration of the wheel at maximum. Chuck knew the pin loads corresponding to these conditions from his previous calculations. Assuming neutral axes in bending as shown in Exhibit 1, he calculated stresses in the wheel at these and at other points using the simple bending relationship, where

$$\sigma = \frac{Mc}{I}$$

σ = bending, or flexural, stress

M = bending moment

c = perpendicular distance from the neutral axis to outermost fiber

I = moment of inertia of the cross section about the neutral axis.

When Chuck carried out the stress computations, he found that his first results for the stress at the root of the slot were about 30% lower than the stresses determined photoelastically. He said, "From the beginning of the beam theory analysis, we thought that the slot in the wheel would cause the root section to behave as a notched beam. So when the computed results were low, we felt justified in introducing a stress concentration factor. This brought the computed results to within 5% of those obtained from the plastic model.'

"I now felt that we had accumulated enough information and knowledge not only to minimize contact stress, hence wear, of the model 'A' card machine geneva but also to optimize a geneva for <u>any</u> application. We had a mathematical model for internal bending stresses which had been experimentally verified; now we could vary any and all of the mechanism parameters and determine both the contact stress at the slot surface (from the usual Hertz equation) and the maximum internal stress. From Johnson's article, we knew that geneva wheel inertia should be 3/2 of the load inertia for minimum contact stress. We could, if we desired, greatly expand the project and arrive at a design procedure and/or a set of design parameters for genevas with varying numbers of slots under any load conditions. Thus, the question remaining before us was simply how to wrap up the investigation."

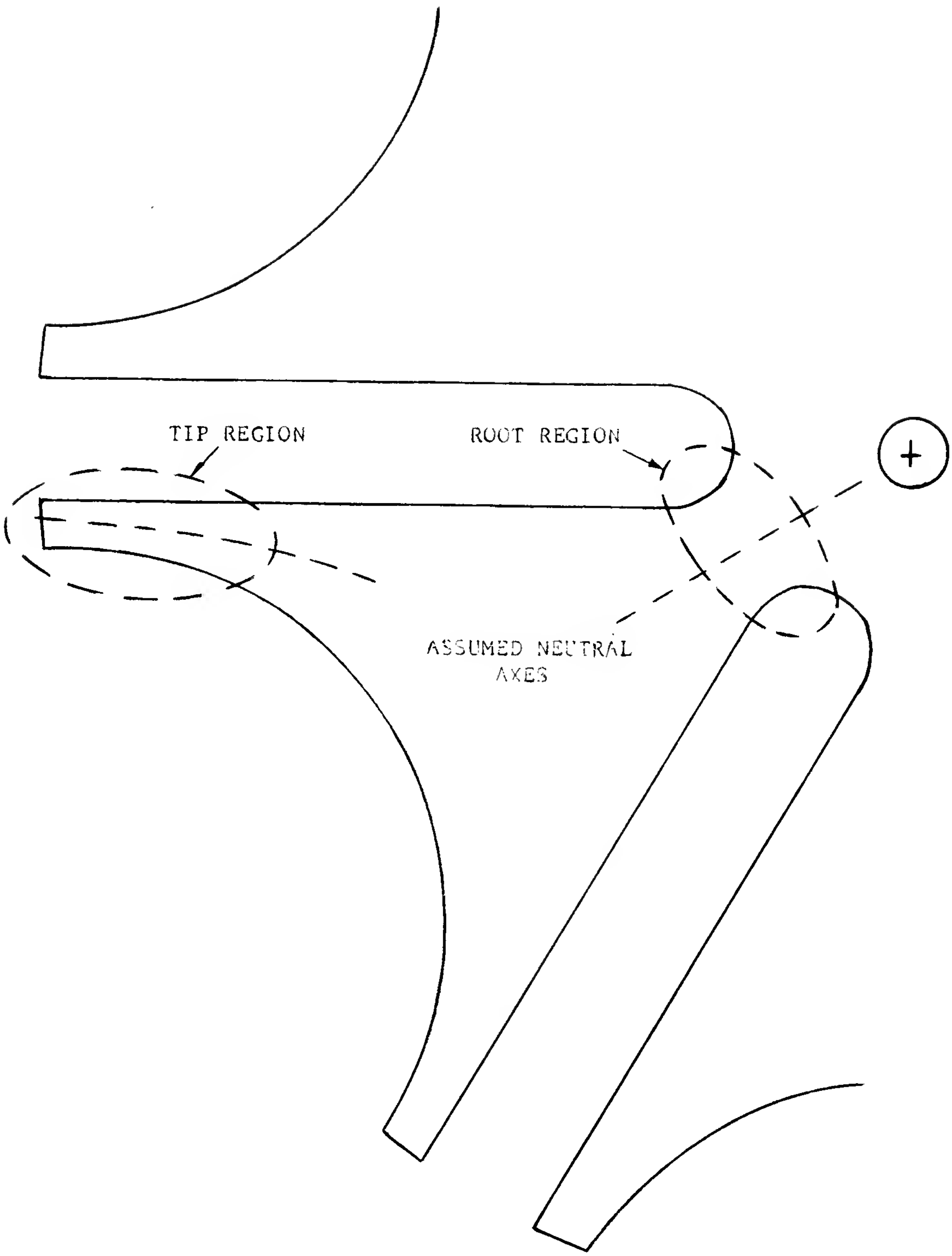


Exhibit 1: Neutral Axes of Geneva Wheel as Observed in Photoelastic Model.

IBM II (D)

Reliability Improvement of a Geneva Mechanism

With a "verified" mathematical model for stresses in the geneva wheel, Chuck Adams felt ready to approach the question of how best to reduce contact stress between pin and slot. His design was to reproportion the star wheel.

Chuck said, "Because of the many variables and the length of computations involved, we decided to write a computer program which would optimize the geneva on the criterion of minimum contact stress. We did this by writing and debugging a number of subroutines and then combining them for the final program. The first subroutine computed the inertia of the star wheel from its dimension and the material density. Another subroutine was developed to compute the force between the pin and slot by the dynamical method we had used before. We wrote tip stress and root stress subroutines and a maximum stress routine to pick out the greatest value of flexural stress. Finally, after preparing a contact stress subroutine upon which to base the optimization of parameters, we assembled the subroutines in a program with appropriate logic to perform the optimization.

"Almost a month was spent preparing and debugging the program. When the first results became available, at the end of February, we sent our recommendations for an improved geneva mechanism to the design group at the San Jose Laboratory." The redesigned mechanism is shown in the drawings of Exhibit 1; it is this design that was shown previously in the photograph of the instrumented driver in Exhibit 1 of Part (B) taken at the time of a further check on the pin loads. The original design and the revised design are compared below:

	ORIGINAL	REVISED
Drive Pin Diameter, in.	.156	.180
Star Wheel Thickness, in.	.125	.180 root; .070 tip
Lock Radius, in.	.292	.280
Star Wheel Diameter, in.	1.1326	1.1326
Pin Load at Entry, lbs.	2.4	2.4
Maximum Pin Load, lbs	13.8	13.8
Root Stress , kpsi	4.9	4.3
Tip Stress, kpsi	2.5	3.9
Maximum Contact Stress, kpsi	42.0	37.2

By March a geneva mechanism of the original design had been on test for 10^7 cycles. An inspection of the star wheel after 5×10^6 cycles had shown slot wear totaling .0001 to .0002 inches. An inspection at 10^7 cycles had shown no further wear. On the basis of these results Chuck believed that the redesigned geneva mechanism would have a satisfactory wear life.

Chuck felt that the conclusions of his analysis would be of widespread interest to design engineers and planned to generalize his computer program to obtain a set of design rules for genevas. He said, "Our first program was intended solely to design a wheel for the new card machine.. To generalize the problem we had to add variables in place of some constants in the program. For instance, we let the number of slots vary instead of being constant at five; we let the material density change instead of being steel. The major and more difficult change, however, came in the logic relations among the subroutines and in the form of the results. We spent a lot of time considering various methods of presenting our results before deciding upon a scheme involving normalized variables and lines of constant stress to represent the three-dimensional stress topology in two dimensions. We felt that this would be the best way to present useful information to the practicing engineer. It took us about a month and a half, spread over a much longer period while working on other projects, to finish the generalization of our analysis."

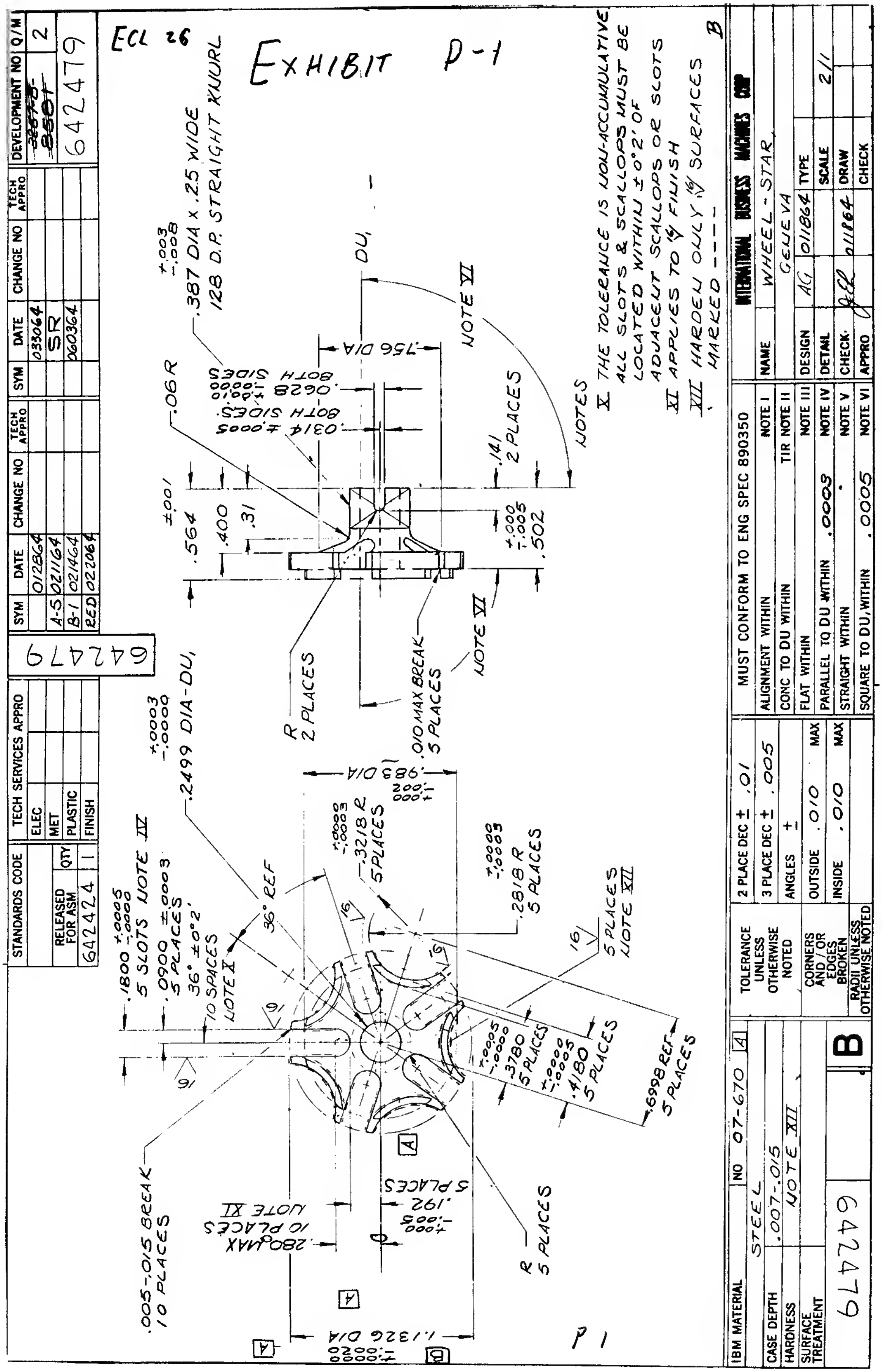


Exhibit 1: Redesigned Geneva Mechanism Incorporating Improved Lock. Sized for Minimum Contact Stress.

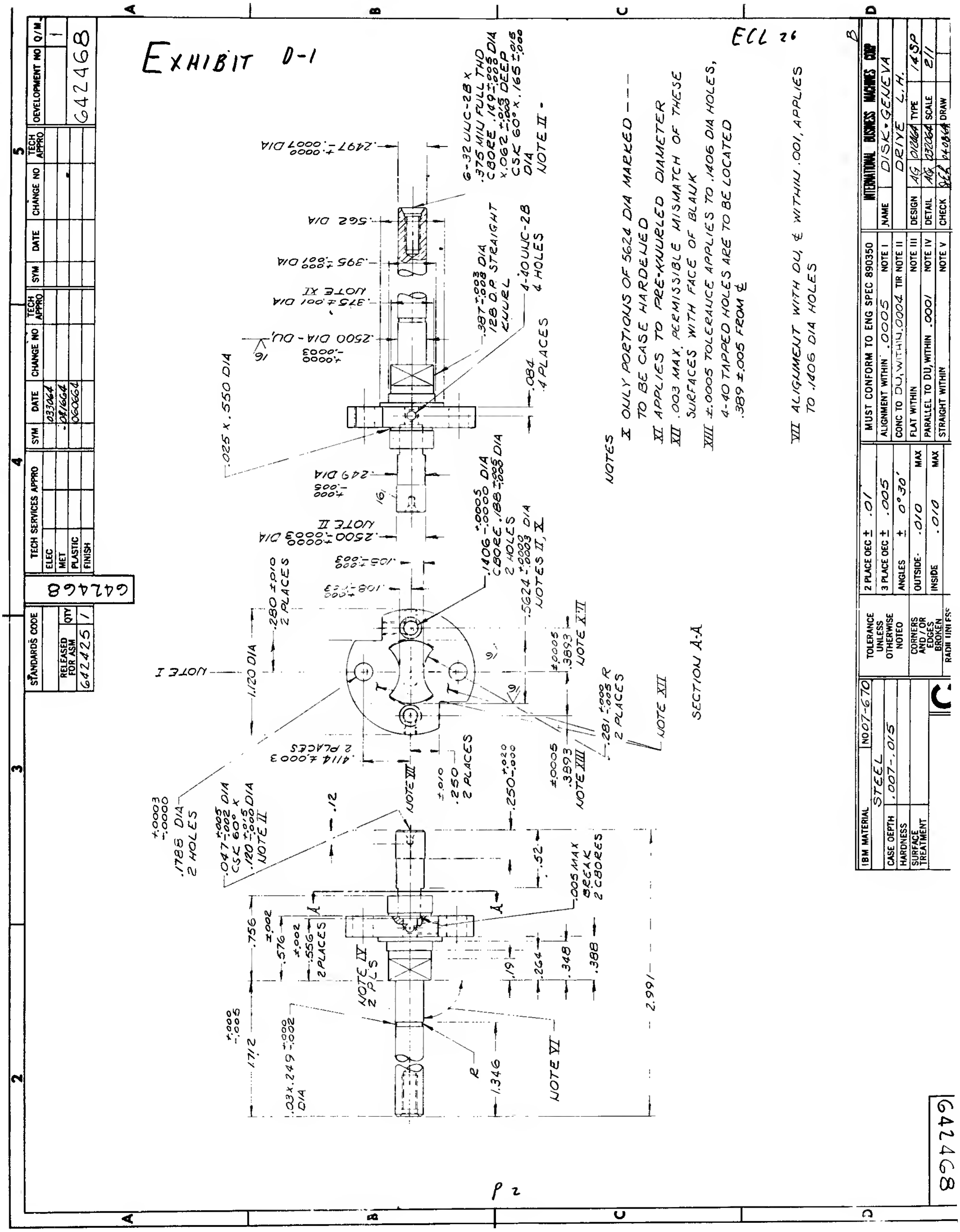


EXHIBIT D-1
ECL 26
NOTES
X ONLY PORTIONS OF 5624 DIA MARKED — TO BE CASE HARDENED
XI APPLIES TO PRE-KNURLED DIAMETER
XII .003 MAX. PERMISSIBLE MISMATCH OF THESE SURFACES WITH FACE OF BLANK
XIII ±.0005 TOLERANCE APPLIES TO .1406 DIA HOLES, 4-40 TAPPED HOLES ARE TO BE LOCATED .389 ±.005 FROM ℄
VII ALIGNMENT WITH DU, ℄ WITHIN .001, APPLIES TO .1406 DIA HOLES
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